

Analysis and Synthesis Procedures for Geneva Mechanism Design

Abstract: This paper contains general analytical results which can be applied to high-speed Geneva design. The results are derived from classical mechanics theory and provide explicit relationships between the performance parameters (those parameters such as contact stress, maximum load, etc., which can have a significant effect on the mechanism performance) and the design variables which specify a Geneva mechanism (number of slots, wheel diameter, pin diameter, etc.). In the past, the complexity of the mathematical formulation of this problem has precluded synthesis of the Geneva wheel proportions. Using these results, however, it is now possible to synthesize the wheel configuration directly, instead of by a repeated trial and error analysis. Two examples are given demonstrating the analysis and synthesis techniques.

Introduction

Geneva mechanisms have long been popular as a means of producing positive incremental motion. This popularity stems in part from the simplicity of the mechanism, both in design and construction, which makes it a relatively low-cost indexing device. In addition, the mechanism inherently produces a precise positioning motion that is necessary for many applications.

In the applications where this mechanism is presently utilized, it has proven to be extremely trouble-free and dependable. In the future it is expected that this device may find many applications requiring higher speeds. As the higher speeds become necessary, the mechanism becomes less attractive as an incremental device because of its kinematic limitations.¹ For instance, a severe limitation under these conditions may result from the high maximum wheel acceleration relative to its average acceleration.² This characteristic may cause excessive dynamic loads which in turn can cause severe drive pin and slot wear and/or wheel breakage.

Therefore the analytical design problem in the case of high-speed Geneva mechanisms, where inertial loads are dominant, is one where the best combination of the design variables is sought to reduce the inherent kinematic limitations of the mechanism.

The primary objective of this paper is to present explicit graphical relationships between the limiting stresses (both wear and breakage) and the available design variables so that their quantitative influence may be readily evaluated by the designer to produce an optimum Geneva design.

These relationships will not only allow one to analyze an existing design but also, more importantly, will allow the designer to synthesize the wheel configuration from maximum stress and/or load criteria.

Design approach

Many factors contribute to a successful Geneva mechanism design, such as materials used, surface finish, tolerances, loads, stress levels, lubricant, etc. Unsuccessful experimental applications of this mechanism usually result in two modes of failure: pin wear and wheel breakage. Of these two modes, wear is the hardest to control. The present design approach will be to reduce wear by altering the geometry of the Geneva wheel to reduce the contact stress while maintaining acceptable stress levels in other regions of the wheel. R. C. Johnson³ showed that an optimum wheel diameter exists for minimum wear stress. In this paper, consideration is given to two additional dimensions (pin diameter and tip thickness) on the wear stress and certain internal beam stresses.

This paper will begin by defining the wheel geometry and then developing the relationships between this geometry and the wheel inertia, the maximum pin load, the contact stress, and the internal wheel stresses. These performance parameters will be normalized to the corresponding parameters of a set of predefined "standard" Genevas for convenience in interpreting results. For the "standard" set chosen, curves will show the stress and load parameters as a function of inertia and speed. The normalized curves

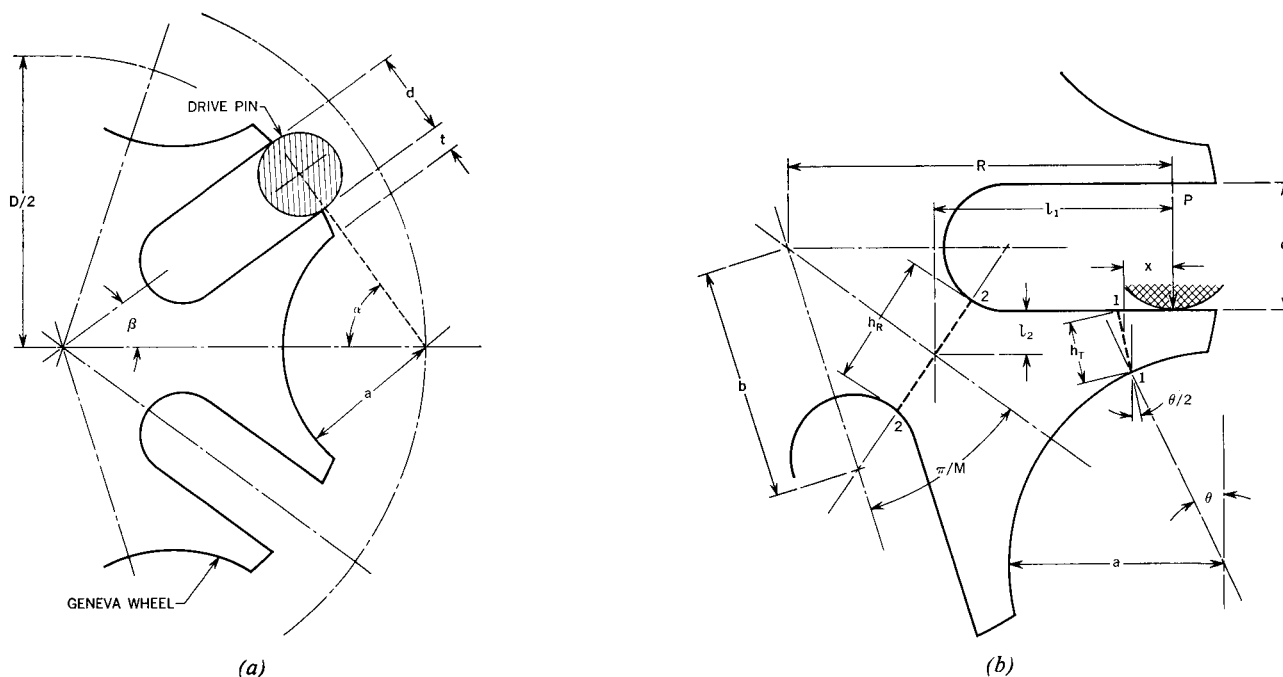


Figure 1(a) Geometry of the Geneva mechanism; (b) beam section geometry.

will show the effect of geometrical differences between any Geneva wheel and the "standard" Geneva.

Graphical curves for 4-, 5-, 6-, and 8-slot Genevas are shown although the concept can be extended to Genevas with any number of slots. The complexity and voluminous nature of the calculations prohibit any complete closed-form solution of the problem, and therefore it was necessary to use a digital computer (IBM 7094) for most of the results. For this reason, no detailed derivations will be given, and the emphasis will be on the results obtained and how they can be used in the analysis and synthesis of Geneva mechanisms.

Analysis

• Wheel geometry

A typical Geneva wheel and drive pin are shown in Fig. 1(a). It is assumed that there is no axial variation in the wheel profile. The three dimensions which specify an M -slot wheel are D , d , and a , i.e., the wheel diameter, drive pin diameter, and lock radius, respectively. It is convenient for our purposes to use an alternate set of dimensions, D , d^* , and t^* , to specify the wheel geometry, where

$$d^* = d/D$$

$$t^* = t/D = \frac{\tan(\pi/M) - d^*}{2} - a/D.$$

For a set of proportional wheels, therefore, the normal-

ized pin diameter d^* and tip thickness t^* will be constant or, conversely, any given d^* and t^* define a proportional set of Geneva wheels. The thickness of the Geneva wheel will not be considered as an independent parameter, but will be taken to be equal to the pin diameter. This particular assumption is made to insure that the drive pin load across the thickness of the wheel is approximately uniform. If the pin diameter is made small with respect to the Geneva wheel thickness, then the loading will be concentrated near the fixed end of the beam and will not be uniformly distributed across the face of the wheel. The final results can be easily modified to include any wheel thickness not equal to the drive pin diameter. This will be demonstrated later in a sample problem.

The six basic design parameters[†] necessary to specify the dynamics of a Geneva mechanism are:

Driver speed	N
Number of slots	M
Load inertia	I_L
Wheel diameter	D
Pin diameter	d^*
Tip thickness	t^*

The remainder of this paper will be directed toward illustrating the effect each of these parameters has on the maximum contact stress, maximum pin load, and maximum internal wheel stresses.

[†] Excluding wheel and pin materials, which are assumed to be steel.

• Wheel inertia

It has been shown³ that the Geneva wheel inertia must be three-halves of the load inertia in order to maintain a minimum pin contact stress. This stress is minimum with respect to the wheel diameter, but, of course, will vary with d^* and t^* . Therefore, if

$$I_G = \frac{3}{2} I_L \sim w D^4 \sim d^* D^5,$$

then

$$D = (3 I_L / 2 K_M)^{0.2}, \quad (1)$$

where K_M is a proportionality constant which depends on d^* , t^* , and M . Thus, the outside Geneva wheel diameter is specified for any I_L , d^* , t^* , and M . This means that the over-all wheel diameter is not to be considered as an independent parameter.

The determination of the diameter consequently will be the last step in the wheel synthesis procedure, after d^* and t^* have been found. Equation (1) can then be used to solve for D . Numerical values of K_M are found in Fig. 2. For notational convenience, the subscript M , which denotes the dependence on the number of slots in the wheel, will be deleted, and it will be assumed that all equations which follow will have this dependence.

• Maximum pin load and contact stress

The maximum normal pin load ($P_{\max}$) occurs when the Geneva wheel is decelerating—a consequence of frictional load. This is given by

$$P_{\max} = \frac{\frac{5}{2} I_L \left(\frac{d^2 \beta}{dt^2} \right)_{\max}}{R_{\max} - \frac{\mu d}{2}}. \quad (2)$$

Let

$R_{\max}^* = R_{\max} / D$, where $R_{\max} = R$ at $P_{\max}$, and

$$\left(\frac{d^2 \beta}{dt^2} \right)_{\max} = \left(\frac{2\pi}{60} \right)^2 \left(\frac{d^2 \beta}{d\alpha^2} \right)_{\max} N^2.$$

It has been assumed in the above that the driver speed is constant. This is usual practice in Geneva design. However, at higher speeds, precautions should be taken to insure that this requirement is satisfied. The maximum value of the second derivative of β with respect to α is a function of the number of slots in the Geneva wheel. The derivation and tabulation of this variable can be found in Ref. 1. Substituting these expressions into Eq. (2) and using the D value shown in Eq. (1) yields:

$$\frac{P_{\max}}{N^2} = \frac{\pi^2 (I_L)^{0.5} \left(\frac{d^2 \beta}{d\alpha^2} \right)_{\max}}{360 \left(\frac{3}{2K} \right)^{0.2} \left(R_{\max}^* - \frac{\mu d^*}{2} \right)}. \quad (3)$$

Since $(d^2 \beta / d\alpha^2)_{\max}$ and $R_{\max}^*$ depend only on the number of slots in the wheel, $P_{\max} / N^2$ can be determined as a function of the given load inertia I_L for any given proportional set of wheels. This relationship is shown in Fig. 3 for a set of wheels which have a d^* and t^* equivalent to those given by Johnson.³ This set of proportional wheels will be referred to as the "standard wheels" denoted by the subscript letter "s." Defining the "standard wheel" was done so that the results can be normalized to some set of stress levels for any load inertia, number of slots, and speed. The values of d^* and t^* for the standard set of wheels are shown in Fig. 3. In Figs. 4(a) through 4(d), the variation of $P_{\max}$ divided by $(P_{\max})_s$ is shown as a function of d^* and t^* where the load inertia, number of slots, and speed remain constant.

The pin-slot contact stress may be computed in a similar way, using Hertz relationships for a cylinder contacting a plane surface:

$$q_0 = \frac{3240}{d^* D} (P_{\max})^{0.5} = \frac{3240 (P_{\max})^{0.5}}{d^* \left(\frac{3 I_L}{2 K} \right)^{0.2}}. \quad (4)$$

The component material is assumed to be steel, with $\nu = 0.3$ and $E = 30 \times 10^6$ lb/in². Using a coefficient of friction of 0.33, it has been shown⁴ that the corresponding maximum contact shear stress, τ , equals $0.43 q_0$.

Thus τ / N can also be determined as a function of the given load inertia, I_L , [using Eqs. (3) and (4)] for a set of proportional wheels. The τ_s / N values for standard wheels are shown in Fig. 3, and values of τ / τ_s are plotted in Fig. 5.

The magnitude of the contact shear stress between two bodies is often used as an indicator of the wear performance to be expected from their combination. Indeed, Bayer et al.⁵ have found empirically that wear life is inversely proportional to the ninth power of the contact shear stress between the two surfaces of interest. This relationship was used to compute the relative wear life of various Geneva designs contained in this report as follows: $L / L_s = (\tau / \tau_s)^{-9}$. Values of this ratio are shown in Fig. 5 on a line which is orthogonal to the lines of constant contact stress. This line can be used as an indicator of the relative wear life for any Geneva defined in the Figure. Values of relative wear life which do not lie on the orthogonal line can be found by moving from the point in question, along a constant τ / τ_s path, to the intersection with the orthogonal line (see illustrative problems).

• Geneva wheel internal stresses

When efforts are made to minimize the pin contact stress by increasing the pin diameter or decreasing the tip thickness, one must consider what effect this will have on the internal stresses of the Geneva wheel. Referring to Fig. 1(b), it can be seen that the load-carrying ability of the wheel

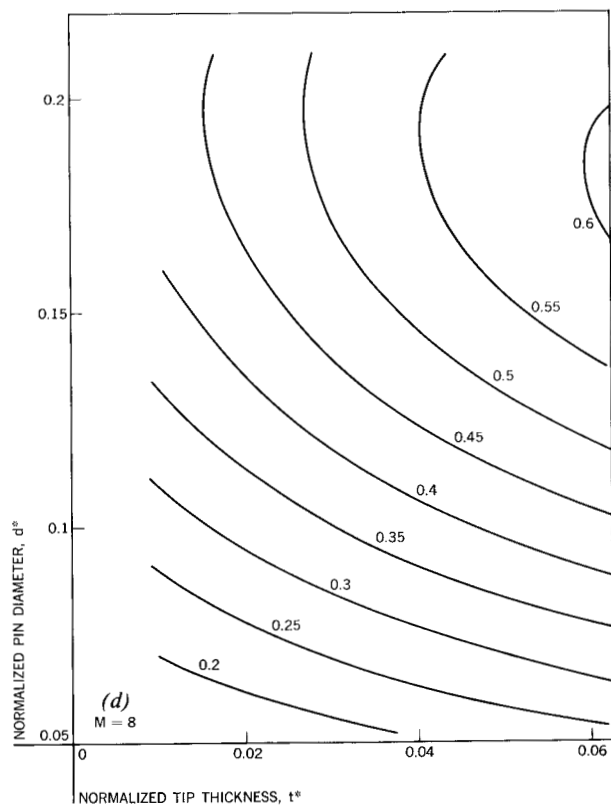
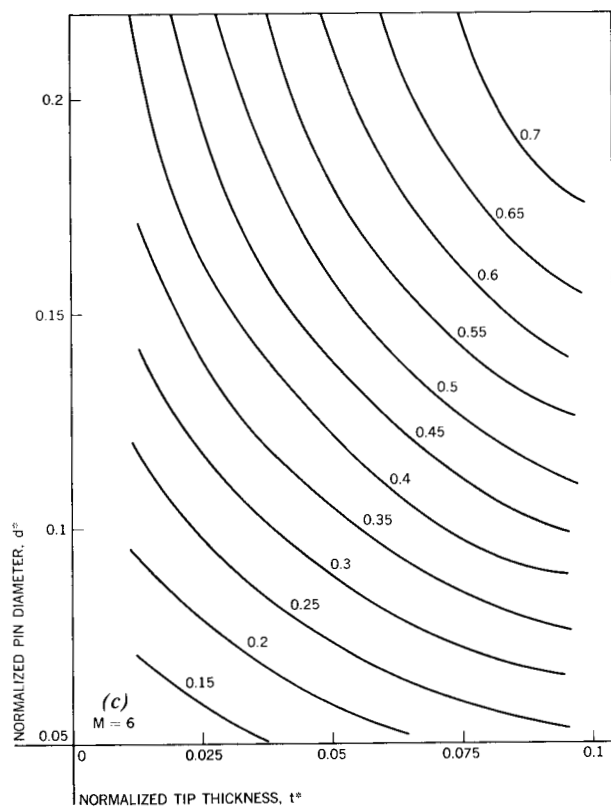
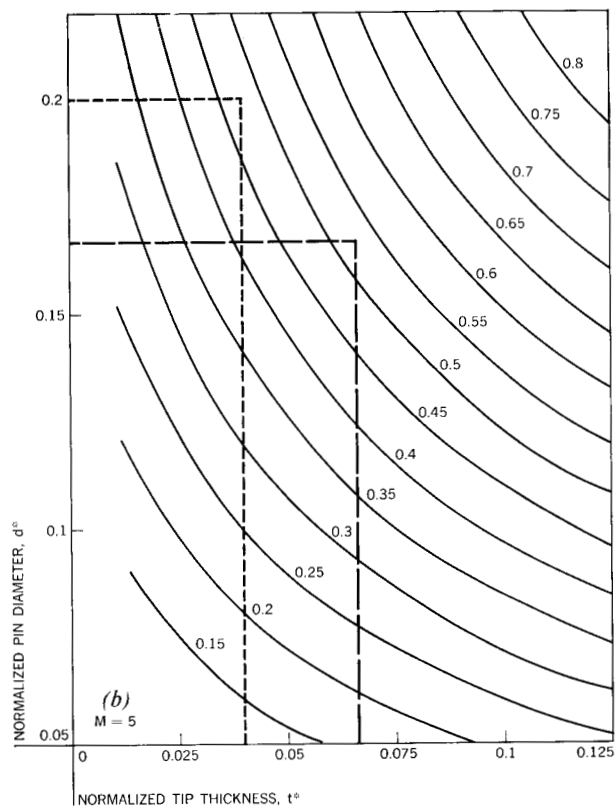
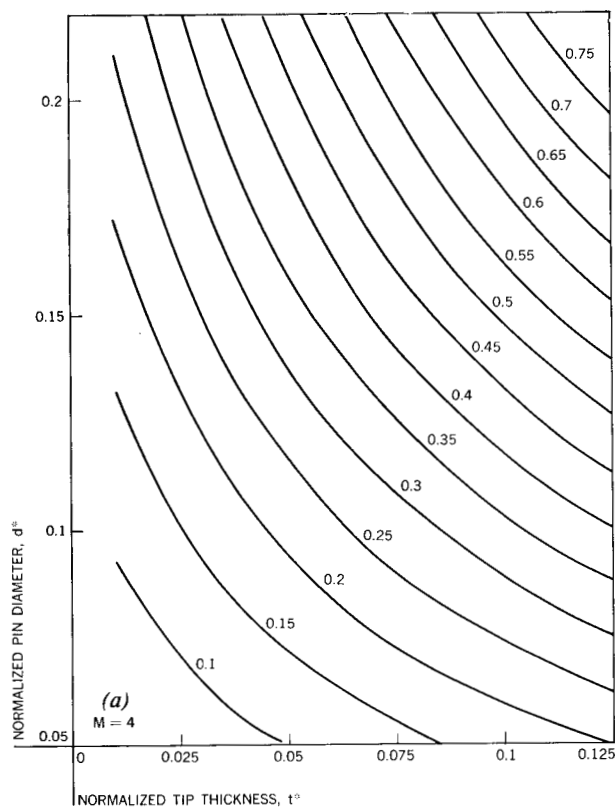


Figure 2 Geneva wheel inertia proportionality constants K_M in $(10^{-5} \text{ lb sec}^2/\text{in}^4)$.

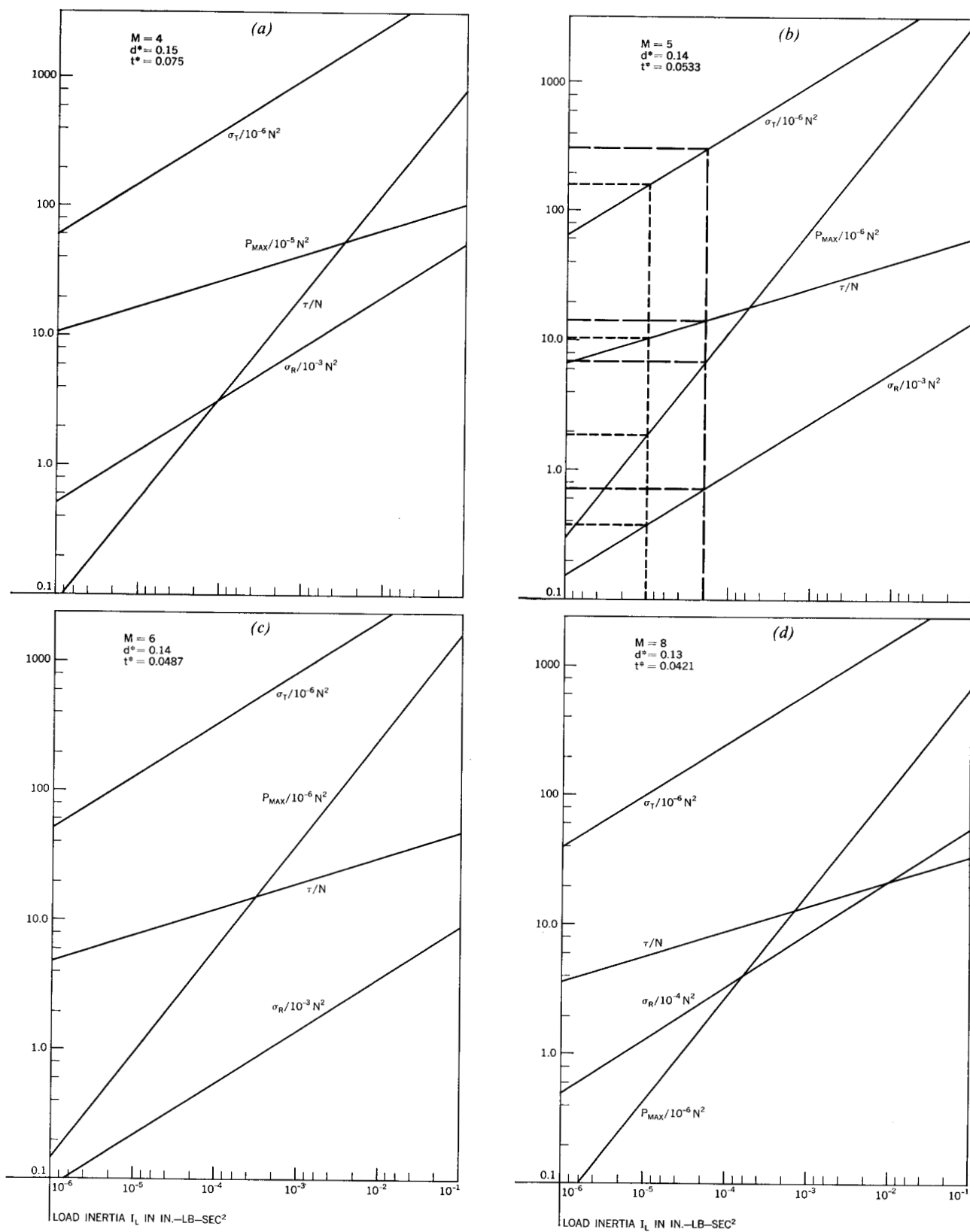


Figure 3 Loads and stresses for the standard Geneva wheel configurations.

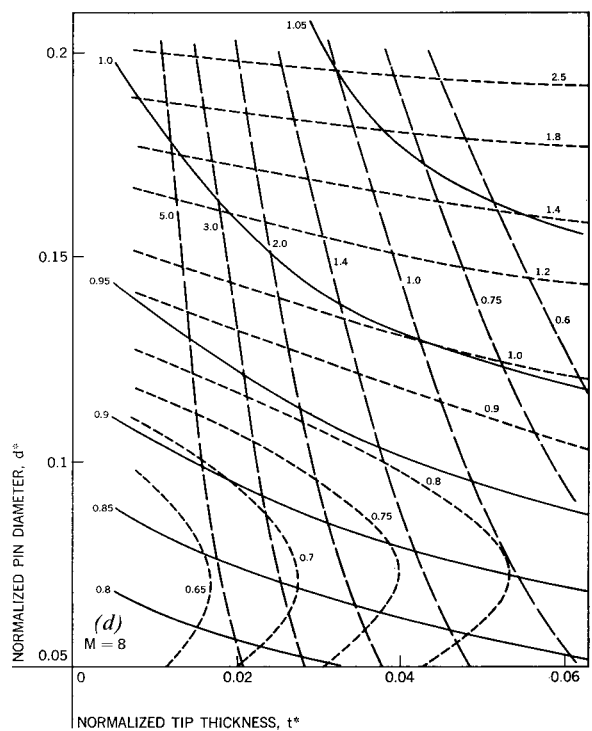
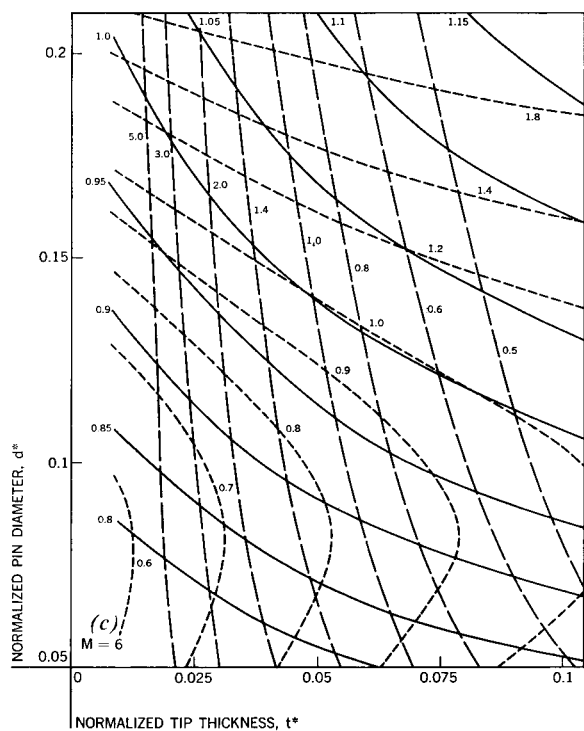
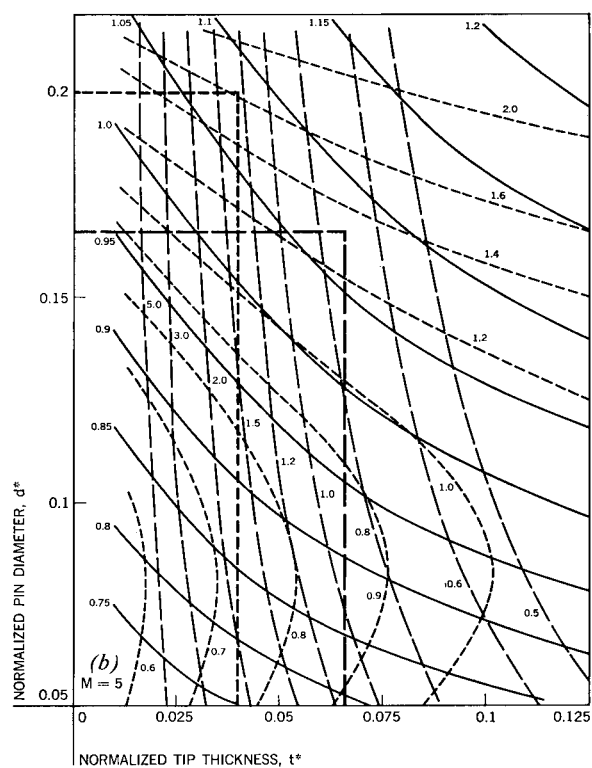
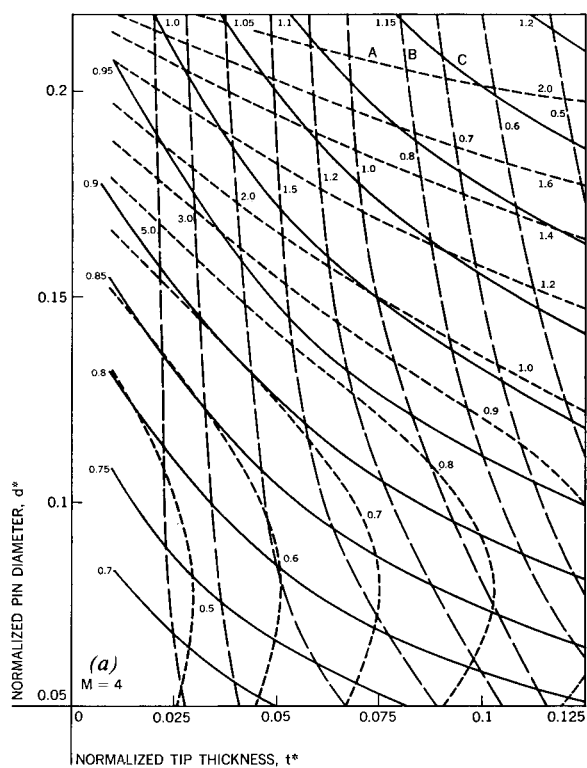


Figure 4 Lines of constant stress and load ratios.

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$$\frac{\sigma_R}{[\sigma_R]S} \text{ --- } \frac{\sigma_T}{[\sigma_T]S} \text{ --- } \frac{P_{\max}}{[P_{\max}]S}$$

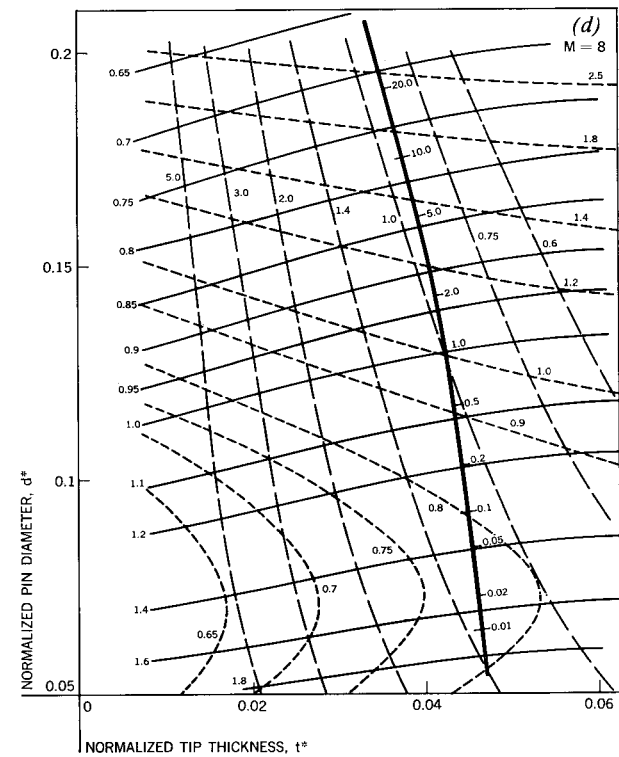
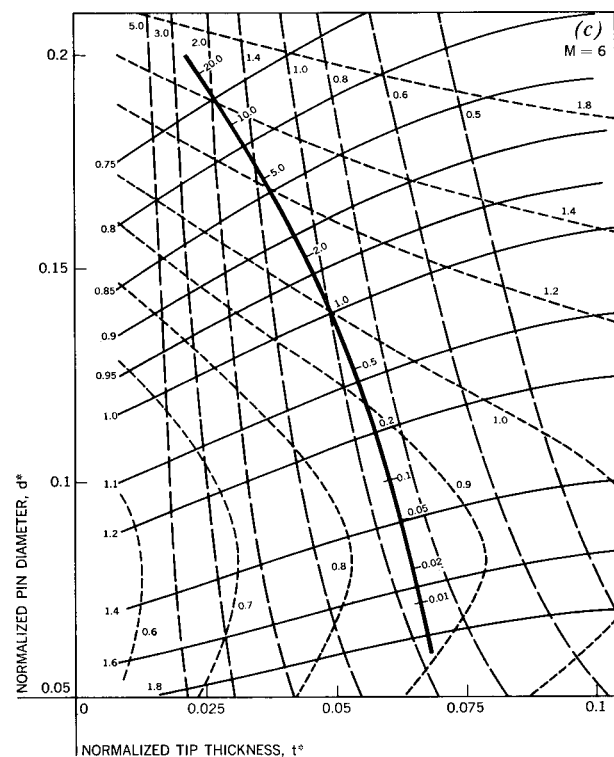
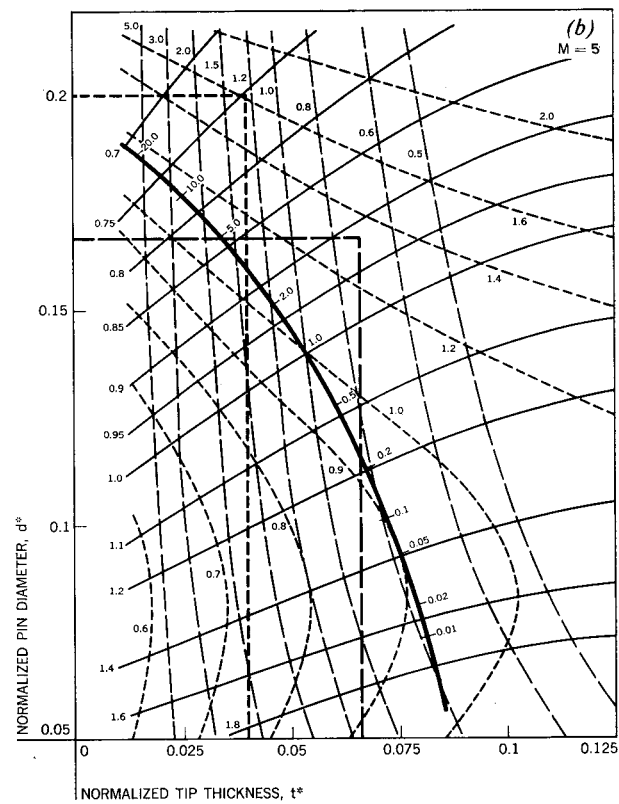
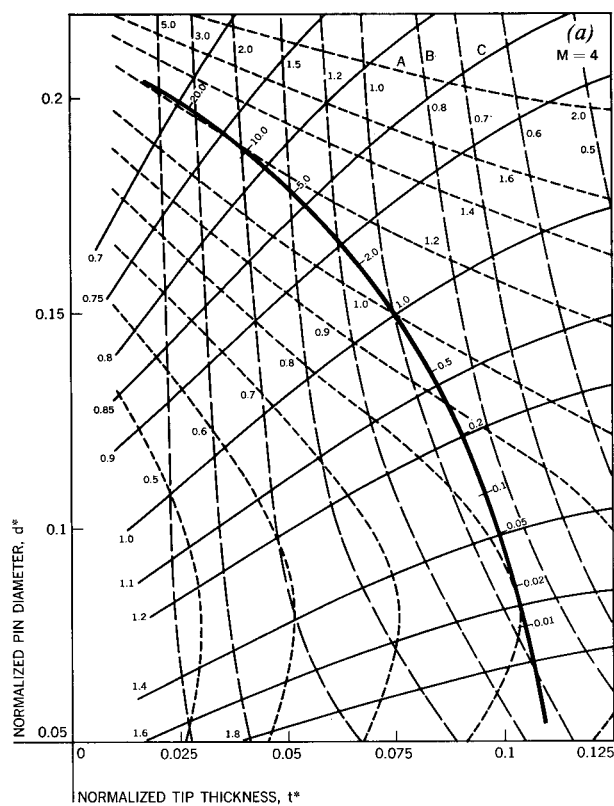


Figure 5 Lines of constant stress and life ratios.

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192 $\frac{R}{R^s}$ ----, $\frac{T}{T^s}$ - · - · -, τ/τ_s —, L/L_s —.

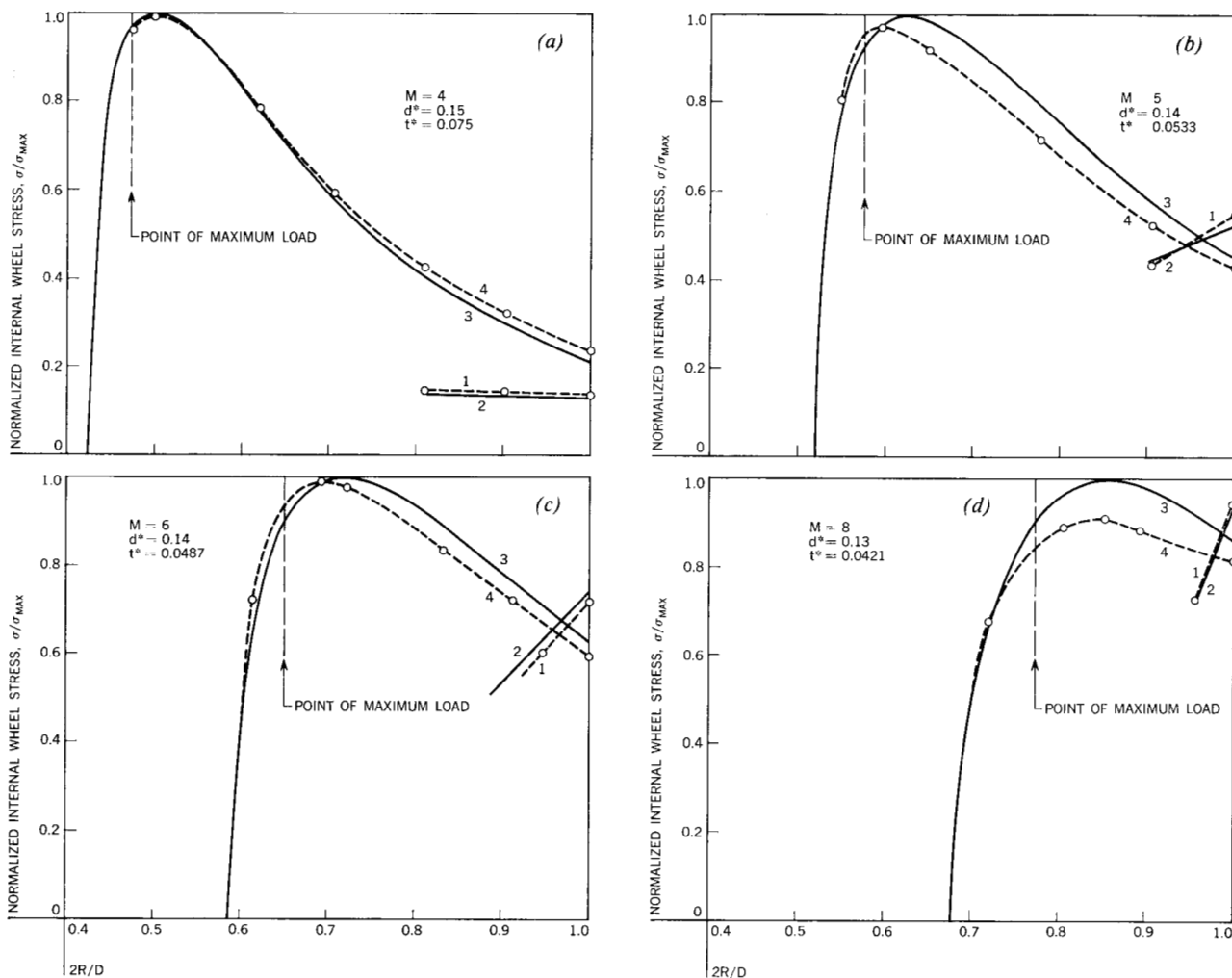


Figure 6 Experimental vs theoretical internal wheel stresses. On each chart, 1 is the experimental tip stress, 2 the theoretical tip stress, 3 the theoretical root stress, and 4 the experimental root stress.

is lost as h_r or h_t approaches zero. For this reason, the stress level in the wheel will be examined as d^* and t^* change. The geometry of the wheel suggests that the maximum stress will occur either at the tip during the initial wheel acceleration (or final deceleration) or at the section near the bottom of the slot.

An approximate determination of these stresses can be afforded by applying beam theory, where these approximate beam sections are denoted by lines 1-1 and 2-2 in Fig. 1(b). The stress produced in the tip during initial pin entry and final pin exit (assuming θ is small) will be

$$\sigma_T = 6Px/wh_t^2 \quad (5)$$

$$\text{where } h_t = (D/2) \tan(\pi/M) - (d/2) - (a^2 - y^2)^{0.5}$$

$$\text{and } y = (D/2) - R + x.$$

After evaluating Eq. (5) to find the maximum value of σ_T with respect to x and P , one can reduce it to

$$\sigma_T = \frac{6 X_0^* P_0}{d^* (h_{t0}^*)^2 \left(\frac{3 I_L}{2 K_M} \right)^{0.4}}, \quad (6)$$

where X_0^* , h_{t0}^* , and K_M depend on d^* and t^* . Using an equation of the form of Eq. (3) for P_0 , and fixing d^* and t^* to the standard values, gives

$$\sigma_T/N^2 = C(I_L)^{0.4}, \quad (7)$$

where C is a constant for each value of M . This equation has been plotted in Figs. 3(a) through 3(d). (The short- and long-dashed lines at the left of Figs. 2(b), 3(b), 4(b) and 5(b) indicate the values used in the sample problems.)

This stress can again be normalized to $(\sigma_T)_s$, and this has been plotted in Figs. 4 and 5. It appears in both figures so that its variation can be seen as one changes the maximum load or contact stress.

Beam theory again was used to compute the stresses across section 2-2 (see Fig. 1(b)). This stress was modified by using stress concentration factors⁶ to account partially for the notch effect resulting from the slots. The resulting root stress is then

$$\sigma_R = \frac{6K_1P(l_1 + \mu l_2)}{wh_R^2} + \frac{K_2P(\sin \gamma + \mu \cos \gamma)}{wh_R} \quad (8)$$

where

$$h_R = 2b \sin \gamma - d$$

$$l_1 = R - b(\cos \gamma)^2$$

$$l_2 = b \cos \gamma \sin \gamma - \frac{d}{2}$$

$$\gamma = \pi/M$$

$$b = D/2[(1 - \sin \gamma)/(\cos \gamma)] - C_L,$$

and C_L is the radial clearance between the drive pin and wheel at the point of maximum slot penetration. This clearance is assumed to be 0.010 inch per inch of Geneva wheel diameter. The stress concentration factors K_1 and K_2 have been taken from circularly notched beams in bending and tension.

Equation (8) can be reduced to the form of Eq. (7), and it has been plotted in Fig. 3. Division by the standard root stress enables one again to express the normalized root stress as a function of d^* and t^* . This has been done in Figs. 5(a) through 5(d).

In order to determine the adequacy of the theory used to compute the internal Geneva wheel stresses, models were made using a birefringent plastic. The stresses were determined photoelastically using a reflecting polariscope with an experimental accuracy of about five percent. Since the beam analysis approximations depend on the wheel proportions, wheels were chosen which had relatively thick root and tip sections. The photoelastic model wheels (4-, 5-, 6-, and 8-slot wheels) each had the proportions of the "standard" wheels.

The models were statically loaded at several load radii and the maximum stress per unit load recorded. Combining this with the pin load as a function of load radius enables one to determine the Geneva wheel stress as a function of the load radius. These data have been plotted in Fig. 6, where they have been normalized to the maximum stress which occurs during the load cycle. Photographs showing three representative load positions are shown in Fig. 7.

From this study, it appears that the simple beam theory is completely adequate for design purposes.



(a)



(b)



(c)

Figure 7 Photographs of photoelastic models of 5-slot Geneva wheel having load position sequences 1, 2, and 3, respectively, in (a), (b), and (c).

Illustrative examples

The following problems are provided to illustrate Geneva synthesis and analysis using results presented in this paper.

• Problem "A"

As an example of wheel synthesis, assume that a 5-slot Geneva is to index a 10^{-5} in-lb-sec² inertial load at a rate of 5000 steps per minute. The drive wheel has two pins, and, therefore, is driven at 2500 rpm. In addition, assume that from material strength considerations, it is necessary to keep the pin slot contact stress below 20,000 lb/in². The following steps indicate one method by which the specified objectives can be reached. Short- and long-dashed lines have been added at the left of Figs. 2(b), 3(b), 4(b), and 5(b) to show how numerical values have been selected for the sample problems. The following information has been given:

$$M = 5 \text{ slots}$$

$$N = 2500 \text{ rpm}$$

$$I_L = 10^{-5} \text{ in-lb-sec}^2$$

$$\tau \leq 20,000 \text{ lb/in}^2.$$

Using Fig. 3(b), the "standard" Geneva mechanism can be seen to have:

$$\sigma_R = 10^{-3} \times 2500^2 \times 0.37 = 2300 \text{ lb/in}^2$$

$$\sigma_T = 10^{-6} \times 2500^2 \times 160 = 1000 \text{ lb/in}^2$$

$$\tau_s = 2500 \times 10.3 = 25,800 \text{ lb/in}^2$$

$$P_{\max} = 10^{-6} \times 2500^2 \times 1.8 = 11.2 \text{ lb.}$$

In addition, by definition, this design has a relative wear life of one. The contact stress τ_s of this design exceeds the specified level, and it will be necessary to ensure that

$$\frac{\tau}{\tau_s} \leq \frac{20,000}{25,800} = 0.775.$$

Choosing τ/τ_s equal to 0.75 and examining Fig. 5(b), one finds many pin diameter-tip thickness combinations that are satisfactory. The choice of any particular point on the τ/τ_s curve requires some decision as to the magnitude of root and tip stress acceptable in the design. Indeed, either of the two stresses may be minimized (at the expense of the other) by moving one direction or the other along the curve. In this example, both stress levels are relatively low, and the decision is not critical. Thus, arbitrarily choosing $d^* = 0.2$ and $t^* = 0.04$, we have [from Figs. 2(b), 4(b), and 5(b)]:

$$\tau = 19,300 \text{ lb/in}^2$$

$$\sigma_R = 1.6 \times 2300 = 3700 \text{ lb/in}^2$$

$$\sigma_T = 1.3 \times 1000 = 1300 \text{ lb/in}^2$$

$$P_{\max} = 1.07 \times 11.2\text{--}12.0 \text{ lb}$$

$$K = 0.48 \times 10^{-5} \text{ lb-sec}^2/\text{in}^4,$$

which satisfies the specified conditions. It should also be noted that the relative wear life is approximately 12 times that of the standard. The mechanism dimensions are

$$D = \left[\frac{3I_L}{2K} \right]^{0.2} = \left[\frac{3 \times 10^{-5}}{2 \times .48 \times 10^{-5}} \right]^{0.2} = 1.26 \text{ in}$$

$$d = d^* D = 0.252 \text{ in}$$

$$t = t^* D = 0.05 \text{ in}$$

$$w = d = 0.252 \text{ in.}$$

• Problem "B"

To illustrate Geneva analysis, assume that the following information is known and that it is necessary to determine the stress-load levels:

$$D = 1.5 \text{ in}$$

$$d = 0.25 \text{ in}$$

$$t = 0.1 \text{ in}$$

$$w = 0.15 \text{ in}$$

$$I_L = 5 \times 10^{-5} \text{ in-lb-sec}^2$$

$$N = 2000 \text{ rpm}$$

$$M = 5 \text{ slots.}$$

The dimensionless parameters are:

$$d^* = \frac{0.25}{1.5} = 0.167$$

$$t^* = \frac{0.1}{1.5} = 0.0667$$

Once again, using Fig. 3(b), the "standard" Geneva has:

$$\sigma_R = 10^{-3} \times 2000^2 \times 0.71 = 2840 \text{ lb/in}^2$$

$$\sigma_T = 10^{-6} \times 2000^2 \times 310 = 1240 \text{ lb/in}^2$$

$$\tau_s = 2000 \times 14 = 28,000 \text{ lb/in}^2$$

$$P_{\max} = 10^{-6} \times 2000^2 \times 6.6 = 26.4 \text{ lb.}$$

However, these performance parameters are for:

$$d^* = 0.14$$

$$t^* = 0.053$$

$$w/d = 1.0$$

and a wheel of optimum diameter. Using Figs. 4(b) and 5(b), with the d^* and t^* of this problem, one finds:

$$\frac{\sigma_R}{(\sigma_R)_s} = 1.3$$

$$\frac{\sigma_T}{(\sigma_T)_s} = 0.7$$

$$\frac{\tau}{\tau_s} = 0.92$$

$$\frac{P_{\max}}{(P_{\max})_s} = 1.08.$$

Therefore

$$\sigma_R = 1.3 \times 2840 = 3700 \text{ lb/in}^2$$

$$\sigma_T = 0.7 \times 1240 = 870 \text{ lb/in}^2$$

$$\tau = 0.92 \times 28,000 = 25,800 \text{ lb/in}^2$$

$$P_{\max} = 1.08 \times 26.4 = 28.5 \text{ lb}$$

$$L/L_s = 2.1.$$

These are performance parameters for a wheel of:

$$d^* = 0.167$$

$$t^* = 0.0667$$

$$w/d = 1.0$$

and of optimum diameter.

The diameter for the wheel above [using Fig. 2(b)] is:

$$D = \left[\frac{3I_L}{2K} \right]^{0.2} = \left[\frac{3 \times 5 \times 10^{-5}}{2 \times 0.53 \times 10^{-5}} \right]^{0.2} = 1.7 \text{ in.}$$

However, the actual dimensions of the wheel are:

$$D = 1.5 \text{ in}$$

$$w/d = 0.6.$$

Several relationships are necessary to translate the performance parameters of above into those corresponding to the analysis wheel. These are given here without derivation, although they can be obtained through manipulation of results presented in the paper:

$$\sigma'_R = \gamma \sigma_R$$

$$\sigma'_T = \gamma \sigma_T$$

$$\tau' = \sqrt{\gamma} \tau$$

$$P'_{\max} = \xi P_{\max} (D/D')$$

where

$$\xi = \frac{I'_G + I_L}{I_G + I_L} = \frac{3 \left(\frac{D'}{D} \right)^5 \frac{w}{d} + 2}{5}$$

$$\gamma = \xi \frac{d}{w} \left(\frac{D}{D'} \right)^3.$$

The prime superscript represents the corrected values.

Using these relationships we obtain:

$$\xi = \frac{3 \left(\frac{1.5}{1.7} \right)^5 0.6 + 2}{5} = 0.591$$

$$\gamma = \frac{0.591}{0.6} \left(\frac{1.7}{1.5} \right)^3 = 1.44$$

$$\sigma'_R = 1.44 \times 3700 = 5330 \text{ lb/in}^2$$

$$\sigma'_T = 1.44 \times 870 = 1250 \text{ lb/in}^2$$

$$\tau' = 1.2 \times 25,800 = 31,000 \text{ lb/in}^2$$

$$P'_{\max} = 0.67 \times 28.5 = 19.1 \text{ lb}$$

$$L/L_s = \left(\frac{28,000}{31,000} \right)^9 = 0.4.$$

These are the performance parameters for the analyzed wheel.

Conclusions

The results presented here certainly do not represent the ultimate procedure in the design of Geneva mechanisms since the analysis avoids any mention of the effects of lubrication, surface finish, material properties, tolerances, etc. However, it is felt that the analytical portion of the design has been significantly aided by the results presented, especially in the area of wheel configuration synthesis. The eventual success of this procedure will depend to a large degree on the validity of the failure criteria used. No extensive tests have been performed over the wide range of parameters covered in this paper. However, the results were successfully applied during the development of an 8000 cycle per minute, 5-slot, 2-pin Geneva driving an inertial load of 5×10^{-6} in-lb-sec². Measurements taken indicated that the rigid body load and constant driver speed assumptions were applicable. The Geneva continued to function properly after approximately one billion index cycles. At this time it appears that the most significant criteria are maximum load (bearing life), maximum contact stress (wear life), and maximum tip and root stress (fatigue life). Thus the results of this paper have been built around consideration of these mechanism parameters.

The most significant limitation of the work is the rigid body assumption used in computing the system dynamics. In cases where this assumption is questionable, then the results must be considered to be approximate. Most of the other assumptions (materials used, friction, etc.) were necessitated only to ensure that the volume of the presented material was kept to a reasonable length.

Symbols

D	Geneva wheel diameter	in
d	Drive pin diameter	in
t	Tip thickness	in
a	Locking radius	in
α	Angular position of drive pin	rad
β	Angular position of Geneva wheel	rad
K_M	Constant of proportionality of Geneva wheel inertia (a function of d^* , t^* , and M)	lb-sec ² /in ⁴
M	Number of slots in wheel	—
I_L	Load inertia	in-lb-sec ²
I_G	Geneva wheel inertia	in-lb-sec ²
N	Angular speed of driver	rpm
R	Distance between wheel center and drive pin center	in
P	Drive pin load (normal component)	lb
q_0	Hertzian normal stress	lb/in ²
w	Thickness of wheel	in
μ	Coefficient of friction	—
τ	Hertzian shear stress	lb/in ²
σ_T	Tip stress	lb/in ²
σ_R	Root stress	lb/in ²
ν	Poisson's ratio	—
E	Modulus of elasticity	lb/in ²

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